

COMPARATIVE PERFORMANCE ANALYSIS OF DIFFERENT TWISTED TAPE INSERTS IN THE ABSORBER TUBE OF PARABOLIC TROUGH COLLECTOR

ATWARI RAWANI, S. P. SHARMA & K. D. P. SINGH

Department of Mechanical Engineering, NIT Jamshedpur, Jharkhand, India

ABSTRACT

Comparative study on performance of different twisted tape inserts in the absorber tube of parabolic trough collector (PTC) are presented in this paper. The heat transfer equations for fully developed flow under quasi-steady state conditions have been developed in order to analyze the thermal characteristics and effect of operating parameters on system performance. The numerical simulations have been employed for five different twisted tape inserts with mass flow rate ranging from 0.06 kg/s to 0.16 kg/s. This study reveals, using of twisted tape insert has great promise for enhancing the performance of PTC. Results show that serrated twisted tape inserts with $x=2$ yields the best performance among all inserts. The Nusselt number with serrated twisted tape insert ($x=2$) is found to be 3.56 and 3.19 times over plain absorber of PTC with mass flow rate of 0.06 kg/s and 0.16 kg/s respectively and the corresponding enhancement in thermal efficiency is 13.63 % and 5.41 % respectively. Also the exergy efficiency has been found to be 15.40 % and 16.46% and the corresponding enhancement factor is 1.12 and 1.05 respectively under conditions.

KEYWORDS: Collector Efficiency Factor, Collector Heat Removal Factor, useful Heat Gain, Efficiency, Entropy Generation & Exergy Efficiency

Received: Dec 05, 2017; **Accepted:** Dec 26, 2017; **Published:** Jan 13, 2018; **Paper Id.:** IJMPERDFEB201871

1. INTRODUCTION

Heat transfer enhancement by the passive technique such as twisted tapes inserts, coil wire inserts, strip, roughened and extended surface are implemented with the expense of additional energy consumption due to the increases in pressure drop. Twisted tape inserted tubes are broadly used in the continuous swirl flow device for enhancing the heat transfer rate in heat exchangers. Many research works on the effect of swirl generator for convective heat transfer enhancements in tubes are available in the literature. Jaisankar et al. [1] experimentally investigated the thermal characteristics of solar water heater with twisted tape inserts of twist ratios 3,4,5 and 6 for the range of Reynolds number, $3000 < Re < 23,000$. They have reported the maximum increase in convective heat transfer coefficient and friction factor with twisted tape of twist ratio 3. Naphon [2] has developed the correlation of heat transfer coefficient for turbulent flow in tube with twisted tape inserts having twist ratio $3.1 \leq H/D \leq 5.5$ and Reynolds number in the range of $7000 < Re < 23,000$. Sharma et al. [3] experimentally investigated the thermal characteristics of Al_2O_3 /water nano fluid in transition flow with twisted tape inserts in tube. Their results shows that enhancement in heat transfer at 0.1% volume concentration of Al_2O_3 /water nanofluid is 23.7% as compared to water and friction factor 1.21 times of water flowing in plain tube. Eiamsa-ard et al. [4] investigated the heat transfer and pressure drop in square duct with tandem wire coil with free space lengths. They found that heat

transfer and pressure drop decrease with increase in free space length. The enhancement in heat transfer rate were obtained by combining several devices with twisted tape inserts in different research work such as corrugated tube [5], conical ring [6], twisted tape with spirally grooved tube [7], dimpled tube [8], twisted tape with wire coil [9] and wire coil tabulators [10]. Chang et al. [11] developed a broken twisted tape for mixing the working fluid and compared with smooth twisted tape insert in the tube and found that the heat transfer coefficient, thermal performance factor and mean fanning friction factor enhanced by 1.28-2.4, 0.99-1.8 and 2.0-4.7 times respectively, for Re of 1000-40000. Manglik and Bergles [12] developed the correlation for Nusselt number and fanning friction factor on the basis of experimental data for water and ethylene glycol with the twisted tape inserts in the tube. Chang et al. [13] analyzed the enhancement in heat transfer in the tubes fitted with serrated twisted tape for different twist ratios. They observed that the fanning friction factor and Nusselt number increases as the twist ratio decreases. Wen and Ding [14] analyzed the enhancement in convective heat transfer coefficient of Al_2O_3 nanofluids. The results show that the heat transfer increases with increase in the Reynolds number and the particle concentration in the base fluid. Heris et al. [15] conducted an experiment on Al_2O_3 /water nanofluid for Reynolds number 700 - 2050 under isothermal wall boundary conditions. The result shows that enhancement in heat transfer with increases in Peclet number and volume concentration of nano particle. Saha and Dutta [16] investigated thermo-hydraulic performance of regularly spaced twisted-tape with multiple twists in the tape module. Results show that the twisted tapes with steadily decreasing pitch perform not as good as than their uniform-pitch counterparts. Waghole et al. [17] experimentally investigated the Nusselt number, efficiency and friction factor in parabolic trough collector with and without twisted tape inserts using silver/ water nanofluid. Result shows that enhancement in efficiency, friction factor and friction factor are 135%-205%, 1.0-1.75 times and 1.25-2.10 times with respectively to plain tube absorber.

The present paper reports the comparative analysis on performance of different types of twisted tape insert in the absorber tube of parabolic trough collector (PTC). The performance of these inserts were investigated analytically using therminol VP-1 as a working fluid and the effect of mass flow rate on various parameters such as collector efficiency factor, collector heat removal factor, rise in fluid temperature, useful heat gain, efficiency, entropy generation and exergy efficiency have been reported and results have been compared with plain tube absorber.

2. THEORETICAL ANALYSIS

Considering a cylindrical parabolic concentrating collector whose concentrator has an aperture ' W_a ' and length ' L_c ' and rim angle ' ϕ_{rim} ' as shown in Figure 1. The absorber tube has an inner diameter d_i and outer diameter d_o and it has a concentric glass cover of inner and outer diameters d_{ci} and d_{co} respectively. The fluid being heated in the collector has a mass flow rate $\dot{m}$, inlet temperature θ_{fi} and an outlet temperature θ_{fo} .

The energy equations under steady state condition can be describe by the following expression for an element of thickness dx of the absorber tube, at distance ' x ' from inlet,

$$dq_u = [I_b r_b (W_a - d_o) \rho \gamma (\tau \alpha)_b + I_b r_b d_o (\tau \alpha)_b - U_L \pi d_o (\theta_p - \theta_a)] dx \quad (1)$$

The left side term in Eq. (1) represents the useful heat gain rate, the first term on right side represents the incident beam radiation absorbed in the absorber tube after reflection, while the second term represents the absorbed incident beam radiation which falls directly on the absorber tube and the third term represents the loss by convection and re-radiation.

The absorbed solar flux ' S ' as follows,

$$S = I_b r_b \rho \gamma (\tau \alpha)_b + I_b r_b (\tau \alpha)_b \left(\frac{d_o}{W_a - d_o} \right) \quad (2)$$

Equation (1), thus becomes

$$dq_u = (W_a - d_o) \left[S - \frac{U_L((\theta_p - \theta_a))}{C_R} \right] dx \quad (3)$$

Where C_R is the concentration ratio of the collector, it is the ratio of effective aperture area to the absorber tube area as,

$$C_R = \frac{(W_a - d_o)L_c}{(\pi d_o)L_c} = \frac{W_a - d_o}{\pi d_o} \quad (4)$$

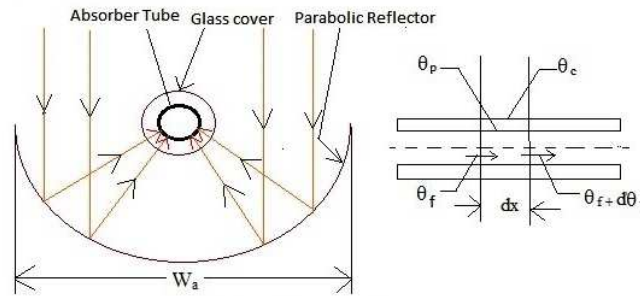


Figure 1: Parabolic Trough Collector

The useful heat gain rate dq_u can be written as

$$dq_u = h_f \pi d_i (\theta_p - \theta_f) dx, \quad (5)$$

Where, h_f and θ_f are heat transfer coefficient on the inside surface of the tube and local fluid temperature. Combining the Eqs. (3) and (5) and eliminating the absorber tube temperature (θ_p), the dq_u can be expressed as;

$$dq_u = F'_c \left[S + (\theta_a - \theta_f) \left(\frac{U_L}{C_R} \right) \right] (W_a - d_o) dx \quad (6)$$

Where F' is the collector efficiency factor defined as

$$F'_c = \frac{1}{U_L \left[\frac{1}{U_L} + \frac{1}{h_f d_i} \right]} \quad (7)$$

Again, combining Eqs. (3) and (5), we obtain the differential equation as

$$\frac{d\theta_f}{dx} = \frac{F'_c U_L \pi d_o}{\dot{m} C_p} \left[\frac{C_R S}{U_L} - (\theta_f - \theta_a) \right] \quad (8)$$

Integrating and using the boundary condition at $x = 0$, $\theta_f = \theta_{fi}$ we have the temperature distribution as,

$$\frac{\theta_f - \left(\frac{C_R S}{U_L} + \theta_a \right)}{\theta_{fi} - \left(\frac{C_R S}{U_L} + \theta_a \right)} = \exp \left[\frac{-F'_c U_L (\pi d_o) x}{\dot{m} C_p} \right] \quad (9)$$

The fluid outlet temperature is obtained by putting $\theta_f = \theta_{fo}$ and $x = L_c$ in Eq.(9). Making this substitution and subtracting both sides of the resulting equation from unity, we have

$$\left[\frac{\theta_{fo} - \theta_{fi}}{\frac{C_R S}{U_L} + \theta_a + \theta_{fi}} \right] = 1 - \exp \left[\frac{-F'_c U_L (\pi d_o) L_c}{\dot{m} C_p} \right] \quad (10)$$

Thus the useful heat gain rate as

$$q_u = F_R(W_a - d_o)L_c \left[S - \frac{U_L}{C_R} (\theta_{fi} - \theta_a) \right] \quad (11)$$

Where F_R is the heat removal factor define by

$$F_R = \frac{\dot{m}C_p}{U_L(\pi d_o)L_c} \left[1 - \exp \left\{ \frac{-F'_c U_L(\pi d_o)L_c}{\dot{m}C_p} \right\} \right] \quad (12)$$

The instantaneous collection efficiency η_i is given as

$$\eta_i = \frac{q_u}{(I_b r_b + I_d r_d) W_a L_c} \quad (13)$$

If ground reflected radiation is neglected. The instantaneous collection efficiency can be calculated on the basis of beam radiation alone, is given by

$$\eta_{ib} = \frac{q_u}{(I_b r_b) W_a L_c} \quad (14)$$

2.1 Overall Loss Coefficient and Heat Transfer Correlations

For calculating the overall coefficient U_L , the correlations are required for calculating individual heat transfer coefficients. The heat loss rate per unit length can be expressed as

$$\frac{q_l}{L_c} = h_{p-c}(\theta_{pm} - \theta_c)\pi d_o + \frac{\sigma\pi d_o(\theta_{pm}^4 + \theta_c^4)}{\left\{ \frac{1}{\epsilon_p} + \frac{d_o}{d_{ci}} \left(\frac{1}{\epsilon_c} - 1 \right) \right\}} \quad (15)$$

$$= h_{p-c}\pi d_o(\theta_{pm} - \theta_c) + \sigma\pi d_{co}\epsilon_c(\theta_{pm}^4 - \theta_a^4) \quad (16)$$

Eqs. (15) and (16) are set of two non-linear equations which have to be solved for the unknowns $\frac{q_l}{L}$ and θ_c after substituting the values of h_{p-c} and h_w .

2.2 Heat Transfer Coefficient between the Absorber Tube and the Cover

The natural convection heat transfer coefficient h_{p-c} for the enclosed annular space between a horizontal absorber tube and a concentric cover is calculated using a correlation by Raithby and Holland [17].

$$\frac{k_{eff}}{k} = 0.317(Ra^*)^{1/4} \quad (17)$$

$$(Ra^*)^{1/4} = \frac{\ln \frac{d_{ci}}{d_o}}{b^{3/4} \left(\frac{1}{d_o^{3/5}} + \frac{1}{d_{ci}^{3/5}} \right)^{5/4}} (Ra)^{1/4} \quad (18)$$

The characteristic dimension used for the calculation of the Rayleigh number is the radial gap, $b = (d_{ci} - d_o)/2$. Properties are evaluated at the mean temperature $(\theta_{pm} + \theta_c)/2$. It can be noted that the effective thermal conductivity k_{eff} can't be less than thermal conductivity k . Hence $\frac{k_{eff}}{k}$ is put equal to unity if the equation (18) yields a value less than unity.

The relation between the heat transfer coefficient h_{p-c} and the heat exchange rate per unit length can be expressed as,

$$\frac{2\pi k_{eff}}{\ln \frac{d_{ci}}{d_o}} (\theta_{pm} - \theta_c) = h_{p-c} (\theta_{pm} - \theta_c) \pi d_o$$

$$\text{Thus, } h_{p-c} = \frac{2k_{eff}}{d_o \ln \frac{d_{ci}}{d_o}} \quad (19)$$

The limitations on using eq. (17) are that Ra^* should be less than 10^7 and b should be less than $0.3d_o$.

2.3 Heat Transfer Coefficient on the Outside Surface of the Cover

The convective heat transfer coefficient h_w on the outside of the cover can be calculated from the well-known correlation based on the data of Hilpert [18] who conducted experiments on air flowing at right angles across cylinders of various diameters at low levels of free stream turbulence. Hilpert's data can be correlated by the equation

$$Nu = C_1 Re^n \quad (20)$$

Where C_1 and n are constant having the following values:

$$\text{For } 40 < Re < 4000, \quad C_1 = 0.615, n = 0.466$$

$$\text{For } 4000 < Re < 40000, \quad C_1 = 0.174, n = 0.618$$

$$\text{For } 40000 < Re < 400000, \quad C_1 = 0.0239, n = 0.805$$

Churchill and Bernstein [19] have made a comprehensive analysis of the data available for cross flow across a cylinder and developed the following correlation

$$Nu = 0.3 + \frac{0.62 Re^{1/2} Pr^{1/3}}{[1 + (0.4/Pr)^{2/3}]^{1/4}} \left[1 + \left(\frac{Re}{282000} \right)^{5/8} \right]^{4/5} \quad (21)$$

Eq.(21) is valid for all values of Re upto 10^7 . For the range $20000 < Re < 400000$, Churchill and Bernstein [19] recommended that the last term $\left[1 + \left(\frac{Re}{282000} \right)^{5/8} \right]^{4/5}$ be modified to last term $\left[1 + \left(\frac{Re}{282000} \right)^{1/2} \right]$ d_{co} is the characteristic dimension to be used in eqs. (20) and (21). Properties are evaluated at the mean temperature $(\theta_c + \theta_a)/2$.

Eqs. (20) and (21) have been obtained for the cross flow and at low level of turbulence intensity. In practice, the flow may not be right angles and the turbulence intensity in the wind may not be insignificant. As a result, there is an uncertainty in the value of h_w predicted by Eqs. (20) and (21). Fortunately, this uncertainty does not affect the value of overall loss coefficient significantly.

2.4 Heat Transfer Coefficient on the Inside Surface of the Absorber Tube

The convective heat transfer coefficient (h_f) on the inside surface of the absorber tube can be calculated under the assumption that the flow is fully developed. For a Reynolds number less than 2000, the flow is laminar and the heat transfer coefficient may be calculated from equation.

$$Nu = 1.86 (Re \cdot Pr)^{0.33} \left(d_i / L_c \right)^{0.33} \quad (22)$$

On the other hand, for a Reynolds number greater than 2000, the flow is turbulent and heat transfer coefficient may be calculated from the well-known Dittus-Boelter equation [20]

$$Nu = 0.023 Re^{0.8} Pr^{0.4} \quad (23)$$

The characteristic dimension used for calculating Nusselt number and Reynolds number in eqs. (22) and (23) is d_i . Properties are evaluated at the mean temperature $(\theta_{fi} + \theta_{fo})/2$. The correlation of the friction factor for plain tube can be

expresses as [23]

$$f_{Pt} = 0.376Re^{-0.259} \quad (24)$$

To enhance the heat transfer, it is desirable to use some kind of augmentative technique as shown in figure. 2 to increase the heat transfer coefficient. One of the simplest techniques is to use a plain twisted tape of width d_i inserts all along the inside absorber tube. The Nusselt number and friction factor [22] for plain twisted tape (PTT) inserts in the PTC tube are evaluated by Eqn. (25) and (26), respectively.

$$Nu = 0.027 Re^{0.862} Pr^{0.33} X^{-0.215} \quad (25)$$

$$f = 2.642Re^{-0.474} X^{-0.302} \quad (26)$$

Where x = twist tape ratio = H/d_i and H = length over which the tape is twisted through 180° .

The Nusselt number and friction factor [22] for square cut twisted tape (SCTT) inserts in the PTC tube are evaluated by Eqn. (27) and (28), respectively.

$$Nu = 0.04Re^{0.826} Pr^{0.33} X^{-0.228} \quad (27)$$

$$f = 6.936Re^{-0.579} X^{-0.259} \quad (28)$$

The Nusselt number and friction factor [23] for oblique delta-winglet twisted tape (ODWTT) inserts in the PTC tube are evaluated by Eqs. (29) and (30), respectively.

$$Nu = 0.18Re^{0.67} Pr^{0.4} \left(\frac{X}{w}\right)^{-0.423} \left(1 + \frac{d_i}{w}\right)^{0.982} \quad (29)$$

$$f = 24.8Re^{-0.51} \left(\frac{X}{w}\right)^{-0.566} \left(1 + \frac{d_i}{w}\right)^{1.87} \quad (30)$$

The Nusselt number and friction factor [24] for alternate clockwise and counter-clockwise twisted tape (C-CCT) inserts in the PTC tube are evaluated by Eqn. (31) and (32), respectively.

$$Nu = 0.31Re^{0.6} Pr^{0.4} \left(X/w\right)^{-0.36} (1 + \sin\theta)^{0.45} \quad (31)$$

$$f = 46.39Re^{0.544} \left(X/w\right)^{0.77} (1 + p\sin\theta)^{0.45} \quad (32)$$

Where, θ = different twist angles.

The Nusselt number and friction factor[25] for serrated twisted tape (SRTT) inserts in the PTC tube are evaluated by Eqn. (33) and (34), respectively.

$$Nu = (0.0118 + 5.84e^{-1.83X})Re^{(0.73-0.695e^{-1.26X})} Pr^{0.33} \quad (33)$$

$$f = (0.033 + 0.756e^{-0.765X})Re^{(0.166-0.235e^{-0.524X})} \quad (34)$$

Table 1: Types of Twisted Tape Inserts used Inside the Absorber Tube of PTC in the Present Investigations

Sl. No	Types of twisted tape	Diagram
1	Plain twisted tape	
2	Square cut twisted tape	
3	Oblique delta-winglet twisted tapes	
4	Alternate clockwise and counter clockwise twisted-tape	
5	Serrated twisted tape	

2.5 Analysis of the Parabolic Trough Collector based on the Second Law of Thermodynamics

In this subsection we developed a model for estimating the Second Law efficiency of the PTC. The set-up is defined by the receiver tube, where different heat transfers occur across the wall-fluid, and the fluid flow. The system comprises dissipative phenomena (or spontaneous non-equilibrium processes) since the natural tendency of system is to achieve equilibrium with their surroundings, and therefore the irreversibility always occur in the actual process.

The overall entropy generation rate S_{gen} of the PTC can be assessed by considering the simpler form [21],

$$\dot{S}_{gen} = \left(\frac{Q_{loss}}{\theta_a} - \frac{Q_s}{\theta_s} + \frac{Q_u}{\theta_{fi}} \right) + \left(\frac{m\Delta P}{\rho\theta_a} \right) \quad (35)$$

Where the first term in parenthesis is due to the heat transfer rate, and the second term is due to the irreversibility caused by the fluid friction. In Eq. (35), θ_a is the ambient temperature and θ_s is the apparent temperature of the sun as an exergy source which is of the order of 4500K. The pressure difference is defined as $-\Delta P > 0$ since there is a pressure drop between the inlet and outlet of the absorber tube. The PTC has an aperture area, A_a , that receives direct solar radiation, G_B , at an energy rate from the sun Q^* as it is shown below in Eq. (36).

$$Q^* = A_a G_B \quad (36)$$

In Eq. (35), the useful heat gain Q_u , is established by Eq.(11), the heat transfer Q_{loss} represents the heat loss to the ambient established by,

$$Q_{loss} = Q^* - Q_u \quad (37)$$

and the pressure drop ΔP can be calculated as,

$$\Delta P = f \frac{4L_c}{d_i} \frac{\rho V_w^2}{2}$$

where L_c is the length of the absorber tube, d_i is the internal diameter of the tube, ρ is the density of the fluid, f is the friction factor and $V_w = m/\rho A$ is the velocity of the fluid.

Rearranging Eq.(25), the entropy generation rate, S_{gen} , of the PTC can be written as,

$$\dot{S}_{gen} = \frac{1}{\theta_a} \left(Q^* \left(1 - \frac{\theta_a}{\theta_s} \right) - Q_u \left(1 - \frac{\theta_a}{\theta_{fi}} \right) + f \frac{32m^3 L_c}{\pi^2 \rho^2 d_i^5} \right) \quad (38)$$

The exergy supplied via solar energy to the PTC is calculated by

$$\dot{E}_s = Q^* (1 - \theta_a/\theta_s) \quad (39)$$

some of this exergy supply is destroyed due to irreversible processes. The exergy destruction E_D of the system is calculated by considering its irreversibility, established by the Gouy-Stodola theorem as,

$$E_D = \theta_a \dot{S}_{gen} \quad (40)$$

and refers to the degraded useful energy when real processes are carried out.

The exergy efficiency η_{TT} is defined as [21]

$$\eta_{TT} = 1 - (E_D / \dot{E}_s) \quad (42)$$

The enhancement factor, Δ_{II} established by considering the second law of thermodynamics as follows

$$\Delta_{II} = \frac{TT}{Pt} \quad (41)$$

3. RESULTS AND DISCUSSIONS

In the following section, results of performance, such as Nusselt number, heat transfer coefficient, collector efficiency factor, collector heat removal factor, rise in fluid temperature, thermal efficiency, entropy generation rate and exergy efficiency of the proposed solar parabolic trough collector are presented. For performing the calculation of this study a computer program in C++ language was developed for the operating and the metrological parameters as given in Table 1. The physical properties including density, the specific heat, the dynamic viscosity and the thermal conductivity of Therminol VP-1 has taken at mean temperature.

Figure. 3 shows, the Nusselt number as a function of Reynolds number for different twisted tape inserts inside the absorber tube of PTC. Nusselt number increases with increase in Reynolds number for constant twist ratio, $X=2$ in all twisted tapes and plain tube. The twisted tapes promote the swirl and turbulence, as a result of which the boundary layer breaks and heat transfer enhances. From figure, it is seen that the maximum Nusselt number is achieved with serrated twisted tape insert as compared to the others twisted tapes and plain tube. The enhancement in Nusselt number is 3.56 and 3.19 times to plain tube at mass flow rate 0.06 kg/s and 0.16kg/s respectively.

Figure.4 represents the plots of the ratio of Nu_{TT}/Nu_{Pt} for different twisted tapes with $H/D = 2$. It is seen that the ratio of Nu_{TT}/Nu_{Pt} of serrated twisted tape is the highest among all twisted tape inserts. It is also found that the ratio of Nu_{TT}/Nu_{Pt} decreases with increasing Reynolds number and this expresses the fact that the twisted tape is better for low turbulence flows.

Table 1: Values of System, Operating and Metrological Parameters

Width of aperture	3 m	Mass flow rate	0.06-0.16 kg/s
Length of absorber	7.8 m	Solar flux	486.03 W/m ²
Inner diameter of absorber	0.0381m	Tilt factor for beam radiation	1.0143
Outer diameter of absorber	0.04315 m	Tilt factor for diffuse radiation	0.993
Glass cover inner diameter	0.0560 m	Slope	9.625 ⁰
Glass cover outer diameter	0.0630 m	Declination angle	9.415 ⁰
Beam radiation	705 W/m ²	Angle of incidence	7.398 ⁰
Diffuse radiation	244 W/m ²	Emissivity/absorptivity of glass	0.88
Intercept factor	0.95	Inlet temperature	120 ⁰ C
Absorber tube Emissivity	0.95	Wind velocity	5.3 m/s
Absorber tube absorptivity	0.95	Ambient temperature	31.9 ⁰ C
Transmissivity	0.85		

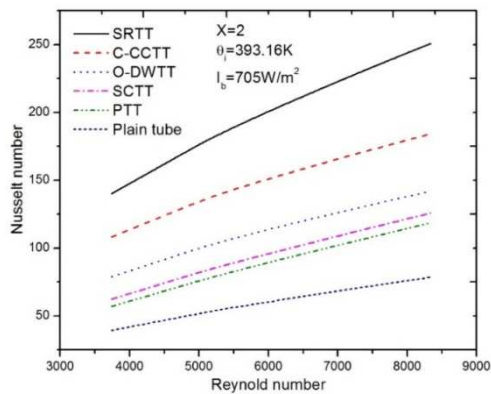


Figure 3: Variation of Nusselt Number with Reynolds Number different Twisted Tape Inserts

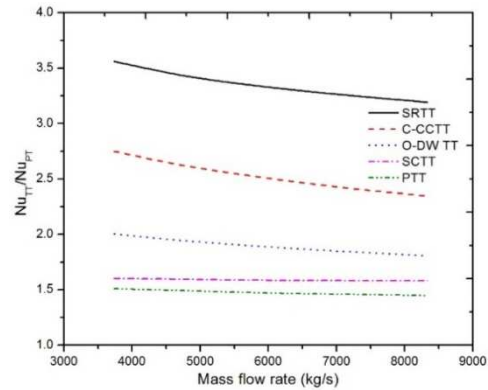


Figure 4: Variation of Nu_{TT}/Nu_{Pt} with Reynolds Number for at Different Twisted Tapes

Figure. 5 shows the variation of heat transfer coefficient as function of mass flow rate for different twisted tape inserts in parabolic trough collector absorber. It is seen in the figure, the heat transfer coefficient increases linearly with increase in mass flow rate for all twisted tapes as well as plain tube. This is due to the fact that increase in mass flow rate increases the Reynolds number and this results higher heat transfer coefficient. The enhancement in heat transfer coefficient with serrated tape inserts as compared to plane tube is found to be 3.56 and 3.192 times at mass flow rate of 0.06 kg/s and 0.16 kg/s respectively.

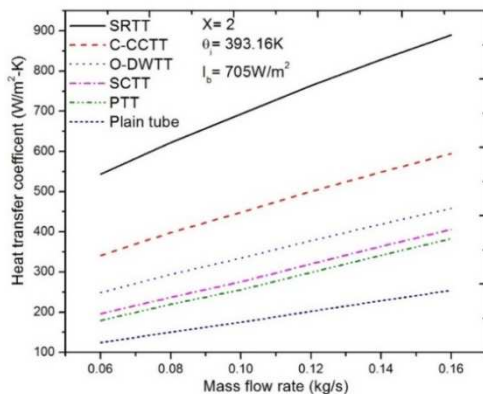


Figure 5: Variation of Heat Transfer Coefficient with Mass Flow different Twisted Tape Inserts

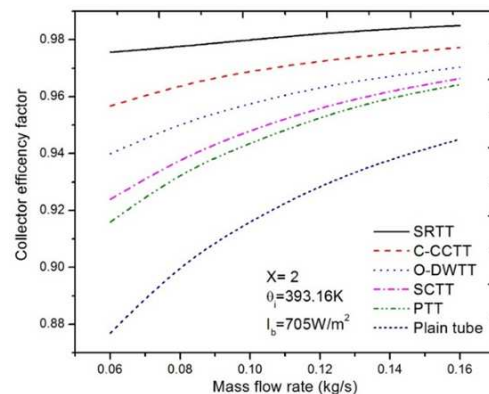


Figure 6: Collector Efficiency Factor as a Function of Mass Rate for Flow Rate for different Twisted Tape Inserts

Figures. 6 and 7 show the variation of collector efficiency factor and collector heat removal factor with mass flow rate of fluid for various twisted tape inserts for $I_b = 705 \text{ W/m}^2$ and $I_d = 244 \text{ W/m}^2$. From the plots, it is seen that collector efficiency factor and heat removal factor increase with increase in mass flow rate of fluid for all twisted tape inserts and plain tube. This is due fact that the heat transfer coefficient increases with increase in mass flow rate consequently increases the collector efficiency factor and collector heat removal factor.

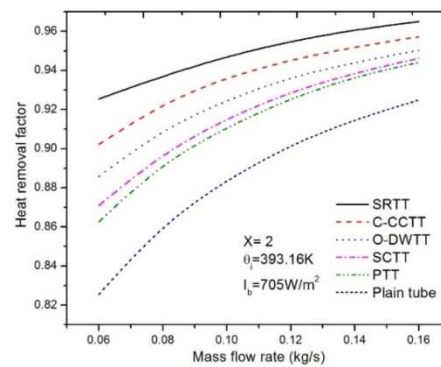


Figure 7: Heat Removal Factor as a Function of Mass Flow

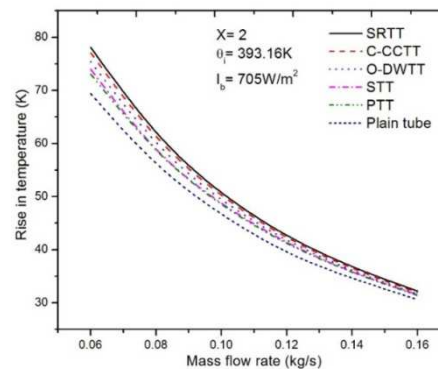


Figure 8: Variation of Rise in Fluid Temperature with Mass Rate at different Twisted Tape Inserts flow rate at different twisted tape inserts

Figure.8 shows that rise in fluid temperature as a function of mass flow rate for different twisted tape inserts at constant twist ratio, $X=2$. The rise in temperature decreases with increase in mass flow rate for all twisted tape inserts as well as plain tube. The maximum rise in temperature is obtained with mass flow rate of 0.06 kg/s and however the minimum rise in temperature is with flow rate, $m=0.16$ kg/s. The maximum rise in temperature is attained at serrated twisted at lower mass flow rate in comparison to other twisted tapes and plain tube.

Figure.9 shows the variation of efficiency with mass flow rate for different twisted tape inserts. It is seen from the figure that the efficiency increases with increase in mass flow rate for different geometries of twisted tape as well as plain tube. This is due to enhanced heat transfer rate at a higher mass flow rate and it is because of heat flow rate between tube surface and working fluid by generating turbulent swirling flow. It has been found that serrated twist tape gives the maximum efficiency for the entire range of mass flow rate. It has been found that the maximum enhancement in efficiency with serrated twisted tape is 13.63% and 5.41% at mass flow rate 0.06 kg/s and 0.16 kg/s respectively as compared with plain tube.

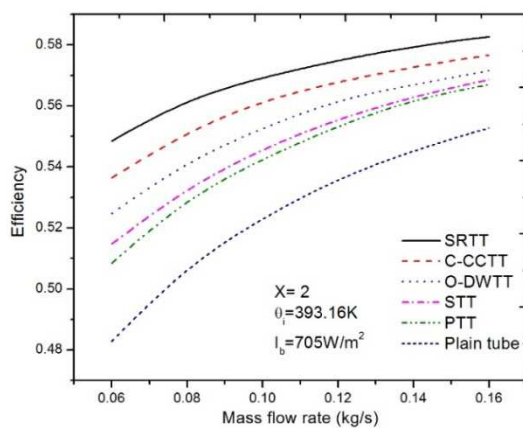


Figure 9: Efficiency as a Function of Mass Flow Rate at Inserts

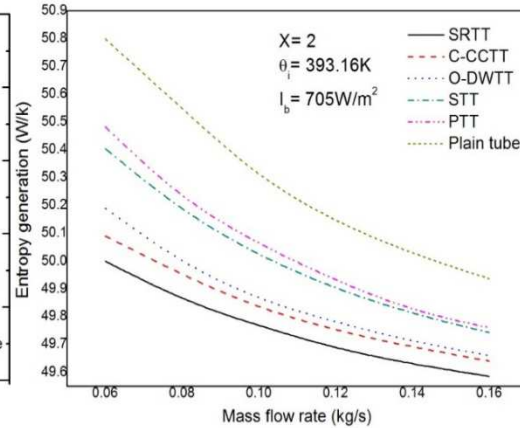


Figure 10: Variation of Entropy Generation with Mass Flow Rate for different Twisted Tape. Rate for different Twisted Tape

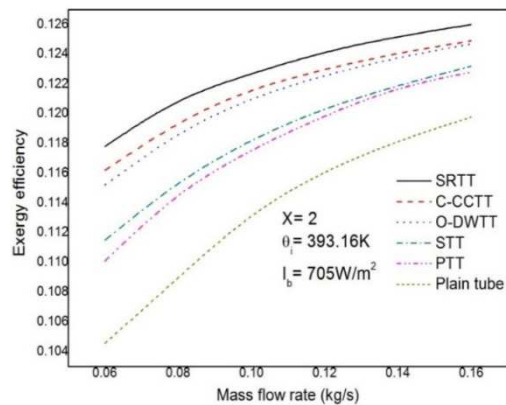


Figure 11: Variation of Mass Flow Rate on Exergy Efficiency for different Twisted Tape Inserts

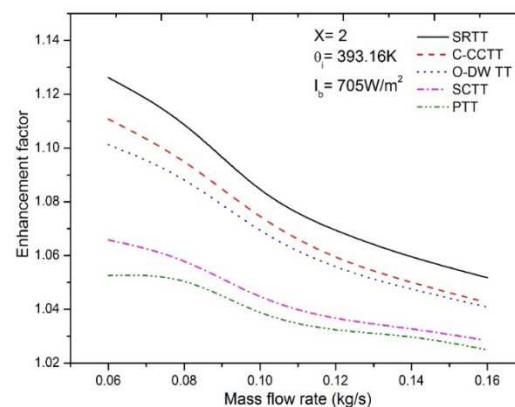


Figure 12: Variation of Ratio of Enhancement Ratio with Mass Flow Rate for different Twisted Tape Inserts

Figure.10 shows the variation of the entropy generation with mass flow rate for different twisted tape inserts with twist ratio, $X = 2$. The entropy generation is calculated from Eq. (38). It can be seen from figure that the entropy generation decreases with increasing mass flow rate. For the specific mass flow rate at constant inlet temperature, the entropy generation rate increases with change of tape twist configurations. It is also seen that the serrated twisted tape insert has lower value of entropy generation rate as compared to other type of tapes at all mass flow rates.

Figure. 11 and Fig.12 show the variation of the exergy efficiency and enhancement factor i.e. for augmentation of exergy efficiency with mass flow rate for different twisted tape inserts. Results of exergy efficiency of plain tube are also plotted for comparison. It is seen from the figure that the exergy efficiency increases and the enhancement factor decreases with increase in mass flow rate. It is also observed from the plot that the enhancement in exergy efficiency i.e. enhancement factor is considerably higher at lower mass flow rate and it is lower at higher mass flow rate. This is due the fact, at higher mass flow rate the entropy generation decreases. It is also seen that the exergy efficiency and enhancement factor increases with change in configurations of twisted tape inserts for entire range of mass flow rate investigated. The maximum exergy efficiency in serrated twisted tape with twisted ratio, $X=2$ is observed to be 15.40% and 16.46% at lower and higher mass flow rate of 0.06 kg/s and 0.16 kg/s respectively and corresponding enhancement factor is 1.12 and 1.05 respectively.

4. CONCLUSIONS

On the basis of above investigations, the following conclusions can be drawn:

- The heat transfer equations have been developed in order to analyze the performance of parabolic trough collector with different twisted tape inserts in the absorber tube.
- A computer program in C++ language is developed in order to study the effect of system and operating parameters on performance.
- The best thermal performance of PTC is achieved for serrated twisted tape insert with $X=2$ among all other twisted tapes inserts investigated.
- It has been found that the heat transfer coefficient increases 3.56 and 3.19 times over plain tube with use of serrated twisted tape inserts ($X=2$) in the absorber tube of parabolic trough collector at mass flow rate of 0.6 kg/s and 0.16 kg/s respectively, while corresponding enhancement in thermal efficiency is found to be 13.63% and 5.41% respectively.
- It has been found that entropy generation decreases with increasing mass flow rate and the lowest entropy generation is obtained with serrated twisted tape.
- The exergy efficiency with serrated twisted tape inserts having tape twist ratio 2, at mass flow rate of 0.06 kg/s and 0.16 kg/s has been found to be 15.40% and 16.46% respectively, while corresponding enhancement factor is found to be 1.13 and 1.05 respectively.

Nomenclature

A_c surface area of collector, m^2 C_p specific heat, J/kg-k C_R concentration ratio dq_u useful heat gain rate E_D exergy destruction $\dot{E}_s$ exergy supply F'_c collector efficiency factor F_R collector heat removal factor f friction factor h_i inside surface heat transfer coefficient, $W/m^2 \cdot K$ h_{p-c} heat transfer coefficient between absorber plate and glass cover, $W/m^2 \cdot K$ h_w wind heat transfer coefficient, $W/m^2 \cdot K$ I_b incident beam radiation, W/m^2 K thermal conductivity, $W/m \cdot K$ K_{eff} effective thermal conductivity, $W/m \cdot K$ k thermal conductivity of fluid, $W/m \cdot K$ H pitch length of twisted tape, m L_c length of collector, m $\dot{m}$ mass flow rate, kg/s Nu Nusselt number Pr Prandtl number θ_a ambient temperature, K θ_c temperature of cover, K θ_f local fluid temperature, K θ_{fi} temperature of fluid at inlet, K	θ_s apparent temperature of the sun, K S incident solar flux, W/m^2 W_a width of aperture, m θ_{pm} avg. temperature of absorber plate, K Q^* solar beam radiation collected by PTC x twist tape ratio Re Reynolds number Ra Rayleigh number r_b tilt factor Ra^* modified Rayleigh number ν kinematic viscosity, m^2/s $\dot{S}_{gen}$ entropy generation <i>Greek letters</i> α absorptivity of absorber ϵ_p emissivity of absorber surface ϵ_c emissivity of cover ϕ_{te} enhancement in thermal efficiency ϕ_r rim angle ρ specular reflectivity of concentrated surface σ Stefan-Boltzmann constant, $W/m^2 \cdot K^4$ τ transmissivity of glass cover η_i instantaneous collection efficiency η_{TT} exergy efficiency of twisted tape η_{Pt} exergy efficiency of plain tube $\Delta\eta_{II}$ enhancement factor <i>Subscripts</i>
---	--

θ_{fo} temperature of fluid at outlet, K θ_p local temperature of absorber tube, K	Pt plain tube TT twisted tape
---	----------------------------------

REFERENCES

1. Jaisankar S., Radhakrishnan T.K., Sheeba K.N., 2009 Experimental studies on heat transfer and friction factor characteristics of forced circulation solar water heater system fitted with helical twisted tapes, *Sol. Energy* 83: 1943–1952.
2. Naphon P., 2006 Heat transfer and pressure drop in the horizontal double pipes with and without twisted tape insert, *Int. Commun. Heat Mass Transf.* 33 (2) 166–175.
3. Sharma K.V., Sundar L. S., Sarma P.K., 2009 Estimation of heat transfer coefficient and friction factor in the transition flow with low volume concentration of Al_2O_3 nano fluid flowing in a circular tube and with twisted tape insert, *Int. Commun. Heat Mass Transf.* 36: 503–507.
4. Eiamsa-ard S., Koolnapadol N., Promvong P., 2012 Heat Transfer Behavior in a Square Duct with Tandem Wire Coil Element Insert, *Chinese Journal of Chemical Engineering* 20,863–869.
5. Wongcharee K., Eiamsa-ard S., 2012 Heat transfer enhancement by using CuO/water nanofluid in corrugated tube equipped with twisted tape, *International Communications in Heat and Mass Transfer* 39,251–257.
6. Promvong P., Eiamsa-ard S., 2007 Heat transfer behaviors in a tube with combined conical-ring and twisted-tape insert, *International Communications in Heat and Mass Transfer* 34 (7) 849–859.
7. Bharadwaj P., Khondge A., Date A., 2009 Heat transfer and pressure drop in a spirally grooved tube with twisted tape insert, *International Journal of Heat and Mass Transfer* 52 (7)1938–1944.
8. Thianpong C., Eiamsa-Ard P., Wongcharee K., Eiamsa-Ard S., 2009 Compound heat transfer enhancement of a dimpled tube with a twisted tape swirl generator, *International Communications in Heat and Mass Transfer* 36 (7) 698–704.
9. Eiamsa-Ard S., Nivesrangsarn P., Chokphoemphun S., Promvong P., 2010 Influence of combined non-uniform wire coil and twisted tape inserts on thermal performance characteristics, *International Communications in Heat and Mass Transfer* 37 (7) 850–856.
10. Promvong P., 2008 Thermal augmentation in circular tube with twisted tape and wire coil turbulators, *Energy Conversion and Management* 49 (11) 2949–2955
11. Chang S.W., Yang, J.S. Liou, 2007 Heat transfer and pressure drop in tube with broken twisted tape insert, *Exp. Therm. Fluid Sci.* 32, 489–501.[12] Manglik, R.M. Bergles A.E.,1993 Heat transfer and pressure drop correlations for twisted-tape inserts in isothermal tubes: part II - transition and turbulent flows, *J. Heat Transf.* 115 (4) 890–896.
12. Chang S.W., Jan Y.J., Liou J.S., 2007 Turbulent heat transfer and pressure drop in tube fitted with serrated twisted-tape, *Int. J. Therm. Sci.* (46) 506–518.
13. Wen D., Ding Y., 2004 Experimental investigation into convective heat transfer of nanofluid at the entrance region under laminar flow conditions, *International Journal of Heat and Mass Transfer* 47 (24)5181–5188.
14. Heris S.Z., Esfahany M.N., Etemad S.Gh., 2007 Experimental investigation of convective heat transfer of Al_2O_3 /water nanofluid in circular tube, *International Journal of Heat and Fluid Flow* 28, 203–210.

15. Saha S.K., Dutta A., *Thermohydraulic study of laminar swirl flow through a circular tube fitted with twisted tapes*, ASME J. Heat Transfer 123 (2001) 417– 427.
16. Waghole D.R, Warkhedkar R.M, Kulkarni V.S, Shrivastva R.K., 2014 *Experimental investigations on heat transfer and friction factor of silver nanofluid in Absorber/Receiver of parabolic trough collector with twisted tape inserts*. Energy Procedia 45,558-567.
17. Hilpert R., 1933 *Warmeabgabe von geheizten drahten und rohren, forsch gebiete ingenieurw*, (4) 220.
18. Churchill S. W. and Bernstein M., 1977 *A correlation equation for forced convection from gases and liquids to a circular cylinder in cross flow*, Journal of Heat Transfer, Trans. ASME, 99,300.
19. Sukhatme S. P., 2007 *Solar Energy*. Tata McGraw-Hill: New Delhi, India.
20. Jaramillo O. A., Borunda M., Velazquez-lucho K. M. and Robles M., 2016 *Parabolic trough solar collector for low enthalpy processes: An analysis of the efficiency enhancement by using twisted tape inserts*, renewable energy, 93,125-141.
21. Murugesan P., Mayilsamy K. and Suresh S., 2010. *Turbulent heat transfer and pressure drop in tube fitted with square-cut twisted tape*, Chinese Journal of Chemical Engineering, 18(4) 609-617.
22. Eiamsa-ard S., Wongcharee K., Eiamsa-ard P., Thianpong C., 2010 *Heat transfer enhancement in a tube using delta-winglet twisted tape inserts*, Applied Thermal Engineering 30,310–318.
23. Eiamsa-ard S., Promvonge P. Eiamsa-ard, C., 2010 *Performance assessment in a heat exchanger tube with alternate clockwise and counter-clockwise twisted-tape inserts*. International Journal of Heat and Mass Transfer 53, 1364–1372.
24. Chang S.W, Jan Y. J., Liou J. S., 2007 *Turbulent heat transfer and pressure drop in tube fitted with serrated twisted tape*, International Journal of Thermal Sciences 46,506–518.